SIMPLIFIED FIRE GROWTH CALCULATIONS

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As part of a fire protection analysis, it is often desirable to estimate the burning characteristics of selected fuels and their effects in enclosures. Also important for many analyses is the estimation of when fire protection devices such as heat detectors or automatic sprinklers will activate for specific fire conditions. Equations are available, based principally on experimental correlations, which permit the user to estimate these effects.

In this chapter, a brief introduction to enclosure fire effects is presented, along with equations that can be evaluated using hand calculators to provide estimates of particular effects. Generally, the equations presented are well documented and are widely used for such estimates. However, the user is cautioned that most of the equations were developed based on data from experiments that were conducted for very specific, and sometimes idealized, conditions. Therefore, some judgment must be exercised when applying these equations to complex conditions occurring in enclosure fires of general interest.

The equations in this chapter are primarily intended to be used in evaluating fire conditions in enclosures during the pre-flashover fire growth period. Most of the methods presented do not apply to fully developed room fires, such as post-flashover conditions. In addition, these shorthand calculations apply only to the room of fire origin and to a single burning fuel package such as a contiguous grouping of combustibles like an upholstered chair and ottoman or a bookcase full of books. More complicated methods are available for multiroom analysis, but they are beyond the scope of this chapter. See Section 11, Chapter 5, "Deterministic Computer Fire Models." Methods to address multiple fuel package involvement are under development but are not yet available.

For some of the effects, more than one equation is presented. In these cases, one equation may be preferred over the other based on the best match of the experimental basis for the equation to the specific case of interest.

Material properties, such as heat of combustion (Δh_c) and stoichiometric air/fuel mass ratio (r_s), are listed in this handbook in Appendix A, "Tables and Charts." For additional information on material properties or the topics introduced in this chapter, refer to Section 3 of *The SFPE Handbook of Fire Protection Engineering*.[1]

All calculations in this chapter are presented in SI units. For U.S. customary units, see Table 11-10A.

TABLE 11-10A. ***Conversion Factors***

To Convert from SI Units	To U.S. Customary Units	Multiply by
Kilograms	lb (avdp)	2.2046226
Kilojoules	Btu	0.948608
Kilowatts	Btu/hr	3414.99
Meters	ft	3.2808399

Also, if T_k is a temperature in degrees Kelvin, then the same temperature in degrees Celsius or Centigrade (T_c) is given by $T_c = T_k - 273.15$. The same temperature in degrees Fahrenheit (T_F) is given by $T_F = 1.8\ T_c + 32$.

ENERGY RELEASE RATE

Calculation procedures for fire effects in enclosures require knowledge of the *energy release rate* of the burning fuel. The term energy release rate is frequently used interchangeably with *heat release rate*, and it is usually expressed in units of kilowatts (kW) and symbolized by $\dot{Q}$.

Currently, no broadly accepted methods exist for prediction of energy release rates soley on basic measurements of material properties. Recent efforts in this area show promise.[2] However, it is expected that generalized methods will not be available for some time. In addition, in any enclosure fire, the actual rate of heat release is dependent not just on the burning fuel, but also on the fire environment, the manner in which the fuel is volatilized, the efficiency of the vapor combustion, and other physical and chemical effects. Therefore, for the immediate future one must rely on available laboratory test data for the specific or similar fuels. In addition, a knowledge of the complete energy release rate history may be required for many situations. This is particularly desirable where the fuel package exhibits unsteady burning. (See Figure 11-10A.) For those cases where only limiting conditions or worst-case analysis is required, it may be reasonable to assume that the fuel is burning at a constant rate, which simplifies the calculation considerably.

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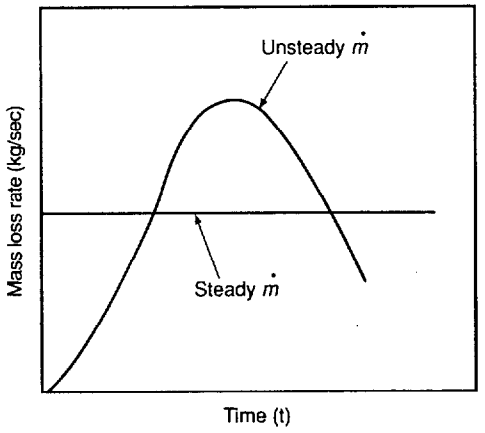

FIG. 11-10A. An illustration of steady and unsteady burning rates.

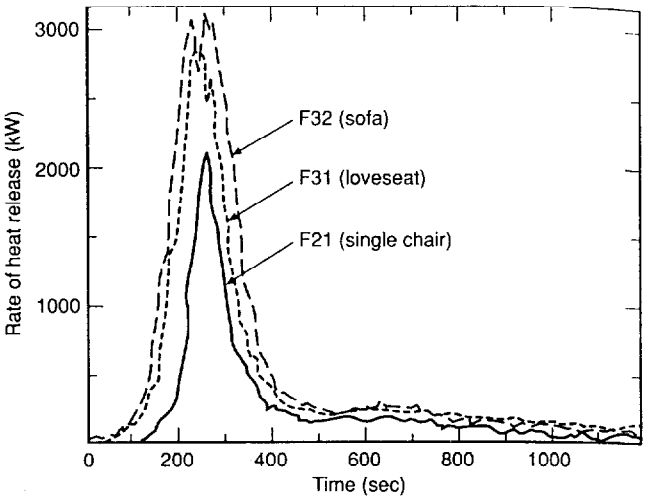

FIG. 11-10B. Free burn heat release rates for selected furniture items.[7]

For the equations presented here, the more simplified condition of constant energy release rate is generally assumed. However, techniques are available that represent a growing fire by a series of constant energy release rate fires. This approach can require a great deal of calculation time, depending upon the desired accuracy. Such analysis is generally more suited to computer simulation.

For the complete combustion of a fuel, energy release rate and mass loss rate are related by the equation:

$$\dot{Q} = \Delta h_c \cdot \dot{m} \tag{1}$$

where

$\dot{Q}$ = energy release rate (kJ/s) or (kW)

Δh_c = heat of combustion (kJ/kg)

$\dot{m}$ = mass loss rate (kg/s)

(The heat of combustion is a material property and is tabulated for selected materials in Appendix A, "Tables and Charts." The mass loss rate is typically found experimentally.)

It should be recognized that most enclosure fires of interest do not exhibit constant energy release rates. Rather, as illustrated in Figure 11-10B for selected items of furniture, the mass loss rate, and therefore the energy release rate, varies over time. Depending upon the detail required, one might select a constant mass loss rate, such as a peak value or an average value as the basis for analysis. Data on mass loss rates for selected fuel packages are available in several publications.[1,3–6]

Most information available on fuel package burning rates is reported for "free burn" conditions—that is, the data are collected for items burning in the open rather than in an enclosure. While enclosure effects are of little importance in evaluating early fire growth, they are important in fully developed room fires. The effects of most importance are those related to radiation feedback to the fuel from the hot smoke and enclosure linings and those related to the ability of the fire to obtain sufficient air for combustion. When fire conditions reach a stage where the smoke and heated room linings approach 932°F (500°C), the radiant feedback normally increases the burning rate above that observed in a free burn situation. The difference between the free burn rate and the radiation-enhanced burning rate increases as the room temperature and resulting radiant impact on the fuel package increase. Once flashover conditions are reached, rates greater than double the free burn rate are not unusual.

The second enclosure effect is the availability of oxygen for combustion. If the air in the space, plus that drawn in through openings, plus that provided to the space by HVAC systems or other means is insufficient to burn all the combustible products driven from the fuel package, only that amount of combustion supportable by the oxygen available in the air will burn within the room or other space involved. This situation is referred to as ventilation-limited burning. When ventilation-limited burning occurs, the combustible products driven from the fuel package and not burned in the room often burn when they combine with air outside the room and appear as flame extensions from the room.

The following equation for stoichiometric fuel pyrolysis can be used to estimate the mass loss rate at which these effects begin to dominate:

$$\dot{m}_{st} = \frac{1}{r_s} \cdot 0.5\, A_v \sqrt{h_v} \tag{2}$$

where

$\dot{m}_{st}$ = stoichiometric mass loss rate (kg/s)

r_s = stoichiometric air/fuel mass ratio

A_v = area of ventilation opening (m^2)

h_v = height of ventilation opening (m)

For wood fuel, $r_s = 5.7$. Values of r_s for other materials can be found in Appendix A, "Tables and Charts."

An estimate of the maximum burning rate possible for an enclosure with a particular opening can be determined from Equation 2. If the mass loss rate for a particular fuel package is less than this value, the condition is referred to as fuel-controlled, and results from Equation 1 provide a reasonable estimate of the energy release rate. If the free burn mass loss rate is higher than the stoichiometric rate from Equation 2, then the rate determined for stoichiometric conditions should be used for combustion within the room.

A more rigorous treatment of energy release rates is available for selected material types such as wood cribs, wood and plastic slabs, and liquid pool fires where experimental correlations have been established. Section 3, Chapter 1, in *The SFPE Handbook of Fire Protection Engineering*[1] provides a detailed discussion of the prediction of burning rates for liquid pool fires. Detailed discussions of energy release rates for specific fuels are available elsewhere.[7,8]

FLAME HEIGHTS

Axisymmetric Flames

Estimates of flame height L can be important in determining exposure hazards associated with a burning fuel. (See Figure 11-10C.) Experimentally determined "mean" flame heights have been correlated by several researchers. A simple correlation for flame heights for pool or horizontal burning fuels has been developed by Heskested:[9]

$$\frac{L}{D} = -1.02 + 15.6N^{1/5} \tag{3}$$

where

L = mean flame height (m)
D = diameter of fire source (m)
N = nondimensional parameter

where

$$N = \left[\frac{C_p T_\infty}{g\rho_\infty^2\left(\frac{\Delta h_c}{r_s}\right)^3}\right]\frac{\dot{Q}^2}{D^5} \tag{4}$$

C_p = specific heat of air at constant pressure [(kJ/kg)K]
$T\infty$ = ambient temperature (K)
g = acceleration of gravity (9.81 m/s^2)
ρ_∞ = ambient air density (kg/m^3)
Δh_c = heat of combustion (kJ/kg)
r_s = stoichiometric air/fuel mass ratio
$\dot{Q}$ = total heat release rate (kJ/s) or (kW)

For noncircular fuel packages, an effective D can be estimated by

$$D = 2\left(\frac{A_f}{\pi}\right)^{1/2} \tag{5}$$

where

D = effective diameter (m)
A_f = area of fire (m^2)

For a broad range of experimental conditions, $\Delta h_c / r_s$ is nearly constant, representing the heat liberated per unit mass of air entering the combustion reaction. Assuming $\Delta h_c / r_s = 3100$ kJ/kg and atmospheric conditions such as $T_\infty = 293$ K, and $\rho = 760$ mm H_g, Equation 3 can be simplified to

$$L = -1.02D + 0.23Q^{2/5} \tag{6}$$

Since flames are unstable, the mean flame height L is generally taken to be the height above the fire source where the flame tip is observed to be at or above this point 50 percent of the time. The above correlation is considered suitable for pool fires or for horizontal surface burning. In addition, the correlation will produce negative values for L at small heat release rates. The available experimental data indicate that the most reliable region of application is where $\dot{Q}^{2/5}/D$ is greater than 16.5. For more detail on flame height calculations, the reader is referred to Beyler[10] and *The SFPE Handbook of Fire Protection Engineering*.[1]

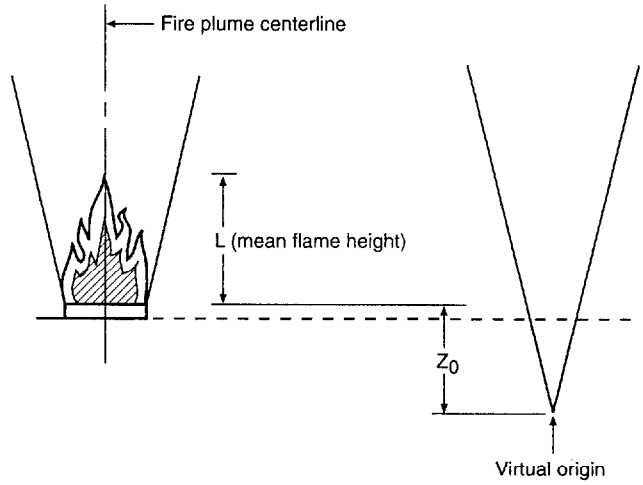

FIG. 11-10C. Flame and fire plume characteristics.

Wall and Line Fires

Equations have also been developed for elongated fires that are either: (1) against a wall so that air is entrained from one side only (i.e., wall fires), or (2) sufficiently in the open so that air is entrained along both of the longitudinal sides (i.e., line fires).[11] In these equations, the flame height is based on the rate of heat release per meter of length of the fire source.

Wall fire flame height:

$$L = 0.034\,\dot{Q}'^{2/3} \tag{7}$$

Line fire flame height:

$$L = 0.017\,\dot{Q}'^{2/3} \tag{8}$$

where

$\dot{Q}'$ = rate of heat release per meter length of the fire source

PLUME CENTERLINE TEMPERATURE AND VELOCITY

The plume centerline excess temperature and velocity at elevations above the mean flame height can be estimated from the following equations.[12] (See Figure 11-10C.)

$$\Delta T_o = 9.1\left(\frac{T_\infty}{gC_p^2\rho_\infty^2}\right)^{1/3}\dot{Q}_c^{2/3}(Z - Z_o)^{-5/3} \tag{9}$$

$$U_o = 3.4\left(\frac{g}{C_p\rho_\infty T_\infty}\right)^{1/3}\dot{Q}_c^{1/3}(Z - Z_o)^{-1/3} \tag{10}$$

where

ΔT_o = excess centerline mean temperature $(T_g - T_\infty)$(K)
T_g = gas temperature (K)

T_∞ = ambient temperature (K)
g = acceleration of gravity (9.81 m/s^2)
C_p = specific heat of air at constant pressure [(kJ/kg)/K]
ρ_∞ = ambient air density (kg/m^3)
$\dot{Q}_c$ = convective heat release rate (kJ/s) or (kW)
Z = elevation above burning fuel fire source (m)
Z_o = location of virtual fire source (m)
U_o = centerline mean velocity (m/s)

Equations 9 and 10 above are based on extensive experimental data and are known as *strong plume* correlations, which accommodate large density deficiencies present in fire plumes. These equations do not apply to fire plumes with small temperature rises, such as $\Delta T_o / T_\infty << 1$.

For normal atmospheric conditions, for example:

$T_\infty = 293$ K
$g = 9.81$ m/s^2
$C_p = 1.00$ [(kJ/kg)/K]
$\rho_\infty = 1.2$ kg/m^3

Equations 9 and 10 can be simplified to:

$$\Delta T_o = A\dot{Q}_c^{2/3}\left(Z - Z_o\right)^{-5/3} \tag{11}$$

$$U_o = B\dot{Q}_c^{1/3}\left(Z - Z_o\right)^{-1/3} \tag{12}$$

where

$A = 25.0$ K m$^{5/3}$ kW$^{-2/3}$

$B = 1.03$ m$^{4/3}$ s^{-1} kW$^{-1/3}$

While methods exist to calculate excess temperature and velocities at locations other than along the plume centerline, the highest confidence is placed on centerline estimates. The reader is referred to DiNenno, Beyler, and Heskestad for a detailed discussion of noncenterline temperature and velocity estimates.[1,10,12]

The use of centerline excess temperature and velocity for evaluating exposure conditions is conservative since the centerline values are the highest values at any elevation. The estimates are sensitive to values for the convective heat release rate— $\dot{Q}_c$ — which can vary from 40 to 80 percent of the total heat release rate, depending on the type and arrangement of the burning fuel.

CALCULATING THE HYPOTHETICAL VIRTUAL ORIGIN (Z_o)

In order to estimate the plume centerline mean temperature and velocity, one must first determine the virtual origin. The virtual origin is the hypothetical location or elevation associated with a substitution of a point source fire for the fire in question. (See Figure 11–10C.) The consideration of virtual origin is most important for evaluating centerline conditions near the fire. As the distance above the fire increases, the impact of the discrepancy that results from neglecting the virtual origin decreases. It is common practice to ignore virtual source considerations for calculations where the distance above the fire is many times the diameter of the fire. For centerline elevations near the fire, however, a more accurate estimate of the position of the virtual source is necessary. The following expression, limited to pool fires and horizontal burning, provides an estimate of the location of the virtual source:[9]

$$Z_o = -1.02D + 0.083\dot{Q}^{2/5} \tag{13}$$

where

Z_o = location of virtual fire source (m)
D = diameter of burning fuel surface (m)
$\dot{Q}$ = total heat release rate (kJ/s) or (kW)

The virtual source can be at, above, or below the base of the burning fuel.

RADIANT HEAT FLUX TO A TARGET

For many enclosure fires, it is of interest to estimate the radiation transmitted from a burning fuel array to a target fuel positioned some distance from the fire to determine if secondary ignitions are likely. Figure 11-10D depicts the configuration used in developing the expression:

$$\dot{q}_0'' \approx \frac{P}{4\pi R_o^2} \approx \frac{\chi_r \dot{Q}}{4\pi R_o^2} \tag{14}$$

where

$\dot{q}_0''$ = incident radiation on the target (kW/m^2)
R_o = distance to target fuel (m)
P = total radiative power of the flame (kW)
χ_r = radiative fraction
$\dot{Q}$ = total heat release rate (kJ/s) or (kW)

Usually, χ_r ranges from 20 to 60 percent, depending upon the fuel type. Refer to Appendix A, "Tables and Charts," for values for χ_r for selected liquid pools. Experimental measurements indicate that Equation 14 has good accuracy for:

$$\frac{R_o}{R} > 4$$

where R (in meters) is the radius of the base of the fire. For radiation at:

$$\frac{1}{2} < \frac{R_o}{R} < 4$$

refer to DiNenno for a more exact analysis.[1]

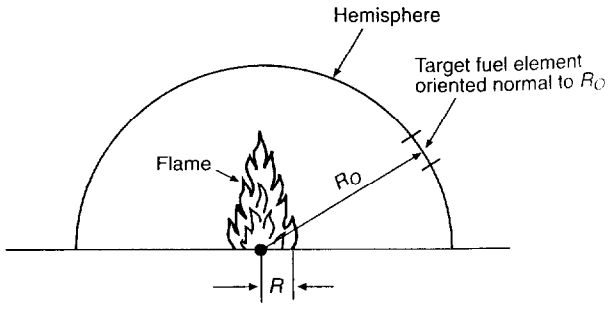

FIG. 11-10D. Illustration of radiant heat transfer to a target fuel.[8]

PRE-FLASHOVER TEMPERATURE ESTIMATES

Several researchers have developed correlations for predicting temperature rise in developing enclosure fires before. McCaffrey *et al.* suggest the following expression for naturally ventilated fires, based on correlation of an extensive number of enclosure experiments:[13]

$$\frac{\Delta T}{T_\infty} = 1.63\left(\frac{\dot{Q}}{C_p \rho_\infty T_\infty A_v \sqrt{g h_v}}\right)^{2/3}\left(\frac{h_k A_s}{C_p \rho_\infty A_v \sqrt{g h_v}}\right)^{-1/3} \quad (15)$$

where

ΔT = temperature rise of upper gas ($Tg - T_\infty$) (K)
T_g = gas temperature (K)
T_∞ = ambient air temperature (K)
$\dot{Q}$ = total heat release rate (kJ/sec) or (kW)
g = acceleration of gravity (m/s)
C_p = specific heat of air at constant pressure [(kJ/kg)/K]
ρ_∞ = density of air (kg/m^3)
A_s = total surface area of enclosure interior excluding vent area (m^2)
A_v = vent area (m^2)
h_v = vent height (m)
h_k = effective enclosure conductance [(kW/m)/K]

The terms $h_k A_s$ and $A_v\sqrt{h_v}$ should be summed in Equation 15 for multiple structural materials and openings, respectively. In addition, while it is recognized that the enclosure gas temperature varies within the compartment, this equation is based on the assumption that an average upper-layer temperature and an average lower-layer, or ambient, temperature reasonably approximate temperature conditions in the enclosure. By substituting values for ambient conditions for key variables in Equation 15:

C_p = 1.0 (kJ/kg)/K, at one atmosphere
ρ_∞ = 1.18 kg/m^3, density of ambient air
T_∞ = 295 K, ambient air temperature
g = 9.81 m/s^2 gravitational constant

A simplified expression can be provided of the form:

$$\Delta T = 6.85\left(\frac{\dot{Q}^2}{h_k A_s A_v \sqrt{h_v}}\right)^{1/3} \quad (16)$$

where

$$h_k = \left(\frac{k\rho c_p}{t}\right)^{1/2} \qquad \text{for } t \le t_p$$

and

$$h_k = \frac{k}{\delta} \qquad \text{for } t > t_p$$

where

k = thermal conductivity of enclosure surface material [(kW/m)/K]
ρ = density of enclosure materials (kg/m^3)
c_p = specific heat of enclosure material [(kJ/kg)/K]
δ = enclosure material thickness (m)
t = time (s)

$$t_p = \frac{\rho c_p}{k}\left(\frac{\delta}{2}\right)^2, \text{ thermal penetration time (sec)}$$

For an enclosure lined with gypsum board, key variable values are

c_p = 1.1 (kJ/kg)/K
ρ = 960 kg/m^3
k = 0.00017 (kW/m)/K
δ = 0.016 m

Thus,

$$h_k = (0.18/t)^{1/2} \text{ for } t \le t_p, (0.00017/\delta \text{ for } t > t_p)$$

where

t_p = 400 sec for 16-mm ($^5/_8$-in.) thickness of gypsum board

For the case of completely forced ventilation conditions, a correlation for enclosure temperature rise has been developed by Foote *et al.*[14]

$$\frac{\Delta T}{T_\infty} = 0.63\left(\frac{\dot{Q}}{\dot{m}_v C_p T_\infty}\right)^{0.72}\left(\frac{h_k A_s}{\dot{m}_v C_p}\right)^{-0.36} \quad (17)$$

where

$\dot{m}_v$ = compartment forced mass ventilation rate (kg/sec)

For detailed discussions of methods that address multiple vents as well as forced ventilation, the reader is referred to DiNenno, Foote *et al.* and Deal and Beyler.[1,14,15]

PREDICTION OF FLASHOVER

A critical point in room fire growth is an event often referred to as "flashover." While a universal definition does not exist, this event is generally associated with rapid transition in fire behavior from localized burning of fuel to involvement of all the combustibles in the enclosure. Experimental work indicates that this transition can occur when upper room temperatures are between 750 and 1112°F (400 and 600°C).[16] Using a value of 932°F (500°C), Equation 15 may be solved for the heat release rate necessary to achieve flashover in a naturally ventilated enclosure.[8] The resulting equation is:

$$\dot{Q}_{fo} = 610\left(h_k A_s A_v \sqrt{h_v}\right)^{1/2} \quad (18)$$

where

$\dot{Q}_{fo}$ = heat release rate at flashover (kJ/s) or (kW)
h_k = enclosure conductance [(kW/m^2)/K]
A_s = total enclosure area (m^2), excluding vent area
A_v = area of vent opening (m^2)
h_v = height of vent opening (m)

Assuming that an enclosure has been heated thoroughly before flashover, i.e., $t > t_p$, Equation 18 can be simplified to:*

$$\dot{Q}_{fo(\min)} = 610\left[(k/\delta)A_s A_v \sqrt{h_v}\right]^{1/2} \qquad (19)$$

where

k = thermal conductivity of enclosure material [(kW/m)/K]
δ_s = enclosure material thickness (m)

POST-FLASHOVER TEMPERATURE ESTIMATES

An alternative approach to estimating peak enclosure temperature was originally developed by Thomas[17] and extended by Law[18] to include both natural and forced ventilation. The correlations were based initially on post-flashover enclosure fire data, but they were extended by Law[18] through the evaluation of extensive pre-flashover room fire data. The results indicate that the predictions reasonably, but not exactly, predict the temperatures reported in the test fires. The equation does not consider variations in the thermophysical properties of room linings.

Most of the tests used to justify the equation involved rooms lined with gypsum board or concrete block. Caution should be exercised in applying these equations to rooms lined with highly insulating materials, such as fiberglass or foamed materials, or rooms in which major portions of the lining are of thermally thin materials such as steel or glass.

Forced ventilation is considered to occur when significant air is supplied by a ventilation system. The natural ventilation equation assumes that, at the time of peak temperature, all of the air for combustion will be drawn into the room through the vent openings and that these same openings will vent the product gases. This results in a natural sharing of the opening by the incoming air and outflowing gases that obey the laws of conservation of mass. The forced ventilation equation assumes that enough air is supplied to ensure that the fire will be free burning. It should be used only when the rate of air supply is sufficient to ensure such burning. Where forced ventilation is not enough to ensure free burning but is sufficient to cause concern that the assumed conservation in the natural venting equation has not been preserved, the peak temperature is expected to lie between the predictions of the two approaches.

The expression for natural ventilation conditions is:

$$\Delta T_{\max} = 6000\left[\frac{\left(1-e^{-0.036\eta}\right)}{(\eta)^{1/2}}\right]\left(1-e^{-0.05\psi}\right) \qquad (20)$$

where

ΔT_{max} = peak temperature rise (K)
η = $[A_s/A_v(h)^{1/2}]$
ψ = $L_f/(A_v A_s)^{1/2}$
A_s = total surface area of enclosure interior excluding vent area (m^2)
A_v = vent area (m^2)
h = height of compartment (m)
L_f = total enclosure fire load (equivalent weight of wood) (kg)

*For 13-mm ($^1/_2$-in.) thickness of gypsum board enclosure liner, $k/\delta = 0.014$. For 16-mm ($^5/_8$-in.) thick gypsum, $k/\delta = 0.011$.

The expression for forced ventilation conditions is:

$$\Delta T_{\max} = 1200\left(1-\theta^{-0.04\psi}\right) \qquad (21)$$

EQUIVALENT FIRE DURATION

Equivalent fire duration or fire severity is an approximation of the potential destructive impact of the burnout of all the available fuel in a room or space with at least one opening. The correlation presented here was developed by Law.[19] The results predict the potential impact of a post-flashover fire in terms of equivalent exposure in a fire-endurance furnace fired to follow the European equivalent exposure of the ASTM E119, *Standard Test Methods for Fire Tests of Building Construction and Materials* (NFPA 251, *Standard Methods of Tests of Fire Endurance of Building Construction and Materials*) standard time-temperature curve. Law based her correlation on data developed through an international research program carried out under the auspices of the Conseil International du Batiment (CIB). The results of this CIB effort are reported by Thomas and Heselden.[20] All of the tests were conducted with wood crib fuel sources. Law reports about 20 percent variation, depending on the porosity of the fuel. In wood cribs, porosity is based on the ratio of openings between the sticks of the crib and the space filled by those sticks, the greater fire severity being experienced with the more loosely packed cribs.

This correlation is not appropriate for rooms that do not have openings for ventilation. While no precise minimum can be stated, it is suggested that this equation not be used unless the area of the opening is at least greater than that of a typical residential window. The equation also assumes that virtually all of the potential energy in the fuel is released in the involved room. This holds true for the wood cribs used in the CIB tests. This assumption may not hold true where there are large surface areas, such as in rooms having combustible linings or in rooms that contain extensive materials with low thermal inertia, such as foam plastics. In these cases, the generation of pyrolized fuel may significantly exceed the combustion ability of the air drawn through the ventilation openings into the fire room. When this occurs, some of the fuel leaves the room in the vented gases. This often burns in the expelled gases causing the extension of flame from the room. In such cases, Equation 22 should be expected to overpredict the equivalent fire duration by an amount approximately proportional to the portion of the fuel that does not burn in the room being evaluated.

$$t = 60\left(\frac{L_f}{\sqrt{A_s A_v}}\right) \qquad (22)$$

where

t = fire severity (s)
A_s = surface area of enclosure interior surfaces, excluding vent area (m^2)
A_v = vent area (m^2)
L_f = wood fuel mass (kg)

SMOKE PRODUCTION RATE

The rate of smoke-filled gas produced by a fire is nearly equal to the rate of air entrained into the rising fire plume, so the mass production rate of smoke-filled gas can be estimated as equal to the mass flow rate of gas in the fire plume. This mass flow rate into a plume above the visible flame height may be estimated using an expression developed by Zukoski:[21]

$$\dot{m}_s = 0.18\dot{Q}^{1/3}\rho_\infty^{2/3}C_p^{-1/3}T_\infty^{-1/3}g^{1/3}Y^{5/3} \qquad (23)$$

where

$\dot{m}_s$ = rate of smoke-filled gas production (kg/s)

$\dot{Q}$ = total heat release rate (kJ/s or kW)

ρ_∞ = density of air (kg/m^3)

C_p = specific heat of air at constant pressure [(kJ/kg)/K]

T_∞ = ambient gas temperature (K)

g = acceleration of gravity (m/s^2)

Y = distance from the virtual point source for the fire to bottom of smoke layer (m)

Assuming an ambient air temperature of 293 K (20°C), Equation 23 can be reduced to

$$\dot{m}_s = 0.065 \dot{Q}^{1/3} Y^{5/3} \tag{24}$$

When the flame height exceeds Y in Equation 24, Equation 24 tends to overpredict gas production.

Equation 24 does not apply to elevations in the flame region. However, McCaffrey[22] has investigated gas temperatures and velocity distributions within the flame and intermittent flame regions for fires up to 250 kW. Under these conditions, the mass flow rate of combustion products was found to correspond to the expression:

$$\dot{m}_s = 0.055 \dot{Q}^{1/2} Y \tag{25}$$

Equations 24 and 25 are based on the assumption that the fire can be reasonably approximated as a circular pool fire. Experiments have shown that reasonable results will be obtained with fires that are not circular, provided that the aspect ratio of length to width is relatively small. The equations are not suitable for conditions where entrainment is restricted (e.g., the fire is against a wall) or if the fire is long and narrow (e.g., a line fire).

An alternative approach to predicting smoke production rates in enclosures has been developed by Butcher and Parnell,[23] based on the size of the fire perimeter and vertical distance to the smoke layer. This approach assumes a constant heat release rate. Based on this approach, smoke production rate can be expressed as:

$$\dot{m}_s = 0.096 P \rho_o y^{3/2} \left(g \frac{T_o}{T_{fl}} \right)^{1/2} \tag{26}$$

where

$\dot{m}_s$ = rate of smoke production (kg/s)

P = perimeter of fire (m)

y = distance from floor to bottom of smoke layer (m)

T_o = ambient temperature (K)

T_{fl} = flame temperature (K)

ρ_o = density of ambient air (kg/m^3)

g = gravitational acceleration (9.81 m/s^2)

Since the expression assumes a constant or steady burning rate, its application has limitations. Yet it will provide a reasonable estimate of smoke generation rate for many enclosure configurations of practical interest.

This expression can be further simplified, based on value assignments for selected parameters. That is, for:

ρ_o = 1.22 kg/m^3 at 17°C
T_o = 290 K
T_{fl} = 1100 K
g = 9.81 m/s^2

Equation 26 is reduced to:

$$\dot{m} = 0.188 P y^{3/2} \tag{27}$$

Figure 11-10E provides graphical results based on the calculation of the smoke-filled gas mass production rate in Equation 26 for selected values of P and y. The mass rate of smoke-filled gas production can be changed to a volume rate by dividing by the density of air at the appropriate gas temperature.

ENCLOSURE SMOKE FILLING

Smoke from a fire begins to fill an enclosure as it accumulates below the ceiling. The rate of smoke filling depends on the amount of smoke produced and the size and location of vents. The mass rate of smoke flow at any distance above a fire of known heat release rate can be calculated using Equation 23. The rate at which a smoke-filled layer descends toward the floor depends on the plan area of the enclosure, the distance of the lower edge of the smoke layer above the fire, and the temperature of the layer.

For an enclosure vented in the lower layer, the upper layer descends with a velocity given by:

$$U_t = \frac{\dot{m}_s}{\rho_l A_p} \tag{28}$$

where

U_t = rate of layer descent (m/s)

$\dot{m}_s$ = mass rate of smoke production (kg/s)

ρ_l = density of smoke layer (kg/m^3)

A_p = enclosure floor area (m^2)

The lower limit for the velocity of descent is obtained by using Equation 28 and the ambient density ρ_∞.

Fires in enclosures in which the upper layer is vented can stabilize at a constant-depth smoke layer. Venting can occur naturally through openings, such as doors and windows, or it can be forced by mechanical smoke control systems.

For the case of a known vent flow rate, the height of the bottom of the smoke layer stabilizes above the fire at the position where smoke mass inflow from the fire plume equals vent outflow. This is calculated using a known vent mass flow rate, $\dot{m}_{sv}$, and solving Equation 24 for position Y_v as:

$$Y_v = 1.9 \dot{Q}^{-1/5} \dot{m}_{sv}^{3/5} \tag{29}$$

This shows that the height at which a smoke layer may be stabilized by venting depends mostly on the vent capacity and is relatively insensitive to changes in the fire heat release rate.

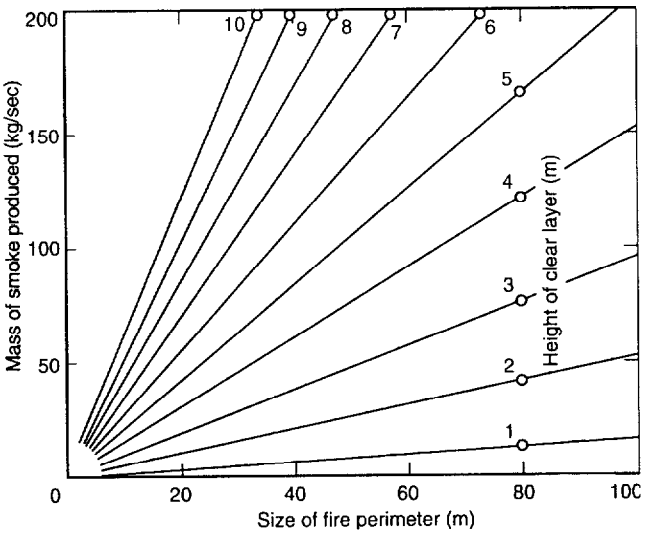

FIG. 11-10E. Smoke production rate for steady fires at various distances (m) from virtual origin to bottom of smoke layer.[23]

BUOYANT GAS HEAD

During a fire, a pressure differential develops between the fire-heated areas and other spaces. Essentially, expressions for pressure differential are derived from the basic hydrostatic equation. The equations below resulted from rearrangement of terms and simplification based on assumed values for selected variables. These expressions are useful in evaluating a range of enclosure effects caused by pressure differentials. Examples include potential for smoke flow to overcome normal air flow, the pressure loading on a door due to the fire, and the uplift pressure on ceiling tiles.

The pressure calculated by these equations results from the difference between the density of a heated gas column from the fire and the density of the surrounding environment. A pressure surge can also result from the expansion of heated gases in those situations where the rate of fire development is fast and the space is not vented sufficiently to relieve the resultant increase in gas volume. The equations given below, however, address only the pressure difference caused by the heated gas column condition.

The equations are universal to any heated gas column and can apply to buoyant gas heads caused by building stack effects or other temperature differential effects, including, but not limited to, those caused by fire. The equations, as presented, assume that the entire column of heated gas is at the same temperature. This is a reasonable approximation in many fires but is not exact and would be inappropriate for a fire where the condition consisted of a plume freely entraining cooling air over an extensive portion of its length or any other condition where a significant temperature gradient existed in the heated gas column being appraised.

The general equation can be expressed as:

$$\Delta P = (\rho_o - \rho_c)gh \tag{30}$$

where

ΔP = pressure difference (Pa)
ρ = gas (air) density outside the heated gas column (kg/m^3)
ρ_c = gas (smoke or flame) density of the heated gas column (kg/m^3)
g = gravitational constant (m/s^2)
h = distance above point where gas column density is same as density outside the heated column. [In a fire, this is normally the base of the hot gas column (m).]

If it is assumed that the outside atmosphere and the gas column are predominantly air at standard atmospheric pressure, the equation can be expressed as:

$$\Delta P = 3460\left(\frac{1}{T_\infty} - \frac{1}{T_c}\right)h \tag{31}$$

where

ΔP = pressure difference (Pa)
T_∞ = absolute temperature of air outside the heated gas column (K)
T_c = temperature of the heated gas column (K)
h = height of the portion of interest of the hot gas column (m)

For further discussion, see Klote and Milke.[24]

THERMAL FIRE DETECTOR RESPONSE

Computer programs have been developed to calculate the response time of heat detectors and sprinklers installed below ceilings in large rooms.[25,26] These programs can determine the time to operation for a user-specified fire energy release rate history. They are convenient to use because the tedious repetitive calculations needed to analyze a growing fire can be avoided. However, the same calculations can be performed easily with a scientific hand calculator for steady fires that have a constant energy release rate. In cases where a more detailed analysis of a fire that has important changes in energy release rate over time is required, the fire may be represented as a series of steady fires occurring immediately after one another.

A useful calculation directly related to thermal detection is to find the plume temperature at positions directly above the flame produced by burning materials. This can be done using the centerline plume temperature correlation or the simplified correlation:[27]

$$T_{m(\text{plume})} = 16.9\frac{\dot{Q}^{2/3}}{h^{5/3}} + T_\infty \tag{32}$$

where

h = distance above fuel surface (m)
$\dot{Q}$ = fire energy release rate [(kJ/s) or (kW)]
$T_{m(\text{plume})}$ = plume gas temperature above fire (K)
T_∞ = ambient room temperature (K)

This equation was developed from analysis of experiments with large-scale fires having energy release rates from 670 kW to 100 MW.[27]

As an example, using Equation 32, the plume gas temperature [$T_{m(\text{plume})}$] 5 m (h) above the fuel surface of a 500 kW ($\dot{Q}$) fire is found to be 366 K (93°C) for an ambient room temperature (T_∞) of 293 K (20°C).

For the case of fixed-temperature detectors, the minimum fire energy release rate, $\dot{Q}$, needed to operate a fixed-temperature detection or suppression device located directly above the fire can be estimated using Equation 32, solving for $\dot{Q}$ with $T_{m(\text{plume})}$ set equal to the activation temperature of the thermal device. In this form, Equation 32 becomes:

$$\dot{Q} = 0.0144\left(T_{m(\text{plume})} - T_\infty\right)^{3/2} \times h^{5/2} \tag{33}$$

Based on cases where the hot gases have begun to spread under a ceiling located above the fire, Equation 33 also applies for a small radial distance, r, from the impingement point. (See Figure 11-10F.) Over this distance, up to $r/h = 0.18$, where the gas is turning to flow out under the ceiling, the highest temperature in the flow remains equal to the value at the impingement point directly over the fire, calculated using Equation 32.

At radial distances greater than $r/h = 0.18$, the maximum temperature in the ceiling jet flow depends upon the distance from the impingement point, according to:

$$T_{m(\text{jet})} = 5.38\frac{\left(\dot{Q}/r\right)^{2/3}}{h} + T_\infty \tag{34}$$

where

h = distance above fuel surface (m)
$\dot{Q}$ = fire energy release rate [(kJ/s) or (KW)]
r = radial distance from plume centerline to device (m),
$T_{m(\text{jet})}$ = temperature of ceiling jet (K)
T_∞ = ambient room temperature (K)

Correlations are also available for maximum velocities in the ceiling jet flow, U_m, under a ceiling. As with the temperature correlations, there are two regions: (1) one close to the impingement point where velocities are nearly constant and (2) the other farther away where velocities vary with radial position. The two correlations are:

$$U_m = 0.96\left(\frac{\dot{Q}}{h}\right)^{1/3} \quad \text{for } r/h < 0.15 \tag{35}$$

and

$$U_m = 0.195\left(\frac{\dot{Q}^{1/3}h^{1/2}}{r^{5/6}}\right) \quad \text{for } r/h > 0.15 \tag{36}$$

where

r = radial distance from plume centerline to device (m)
U_m = gas velocity (m/s)

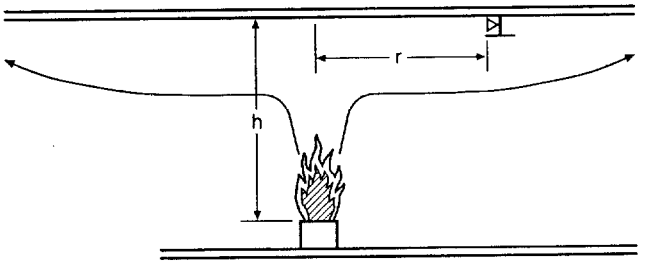

FIG. 11-10F. Parameters h and r both related to calculation of sprinkler or heat-detector actuation time.

The previous equations can be used to determine whether the temperature of the fire-driven gas flow past a detection device is high enough to operate the device. However, more information is needed to calculate the amount of time needed to heat the detector or sprinkler sensing element to the operating temperature. Often, these elements are made of metal, such as the ordinary solder-type fusible link used in the link and lever sprinklers, or they are liquid-filled glass vials, such as those used in bulb-type sprinklers. Both of these sensing elements require some time to absorb the heat transferred from the hot gas flowing around the device.

For steady fires, the time required to heat the sensing element of a thermal detection or suppression device from room temperature to operation temperature is given by:

$$t_{\text{operation}} = \frac{\text{RTI}}{\sqrt{\text{U}_\text{m}}}\log_e\left(\frac{T_m - T_\infty}{T_m - T_{\text{operation}}}\right) \tag{37}$$

where RTI, the response-time index, is a measure of the ease of heating thermal elements in heat detectors and sprinklers. The larger the RTI value, the slower the response of the sensing element. RTI values for sprinklers have been measured[28] in the range of 15 $m^{1/2}s^{1/2}$ to 400 $m^{1/2}s^{1/2}$.

In the previous example, it was found using an earlier equation that a 500 kW $(\dot{Q})$ fire would produce a gas temperature of 366°K $[T_{m(\text{plume})}]$ at 5 m (h) above the fuel surface in a room with 293°K (T_∞) ambient temperature. From Equation 35, the gas velocity at this position would be 4.4 m/s (U_m). For a sprinkler with an RTI of 200 $m^{1/2}$ $s^{1/2}$ and operation temperature of 347°K $(T_{\text{operation}})$, the time to operation in response to the steady fire can be calculated from Equation 37 as:

$$t_{\text{operation}} = \frac{200}{\sqrt{4.4}}\log_e\left(\frac{366-293}{366-347}\right) = 128 \text{ sec.}$$

The measurement of thermal lag for sprinkler is a topic of current research investigations and verification testing.[29,30] It has been found that, for cases when the gas temperature does not substantially exceed the activation temperature for the sprinkler, significant error can occur in the prediction for activation time. In these cases it is possible for small changes in predicted gas temperatures to result in large changes in predicted activation time.[29] In the case of constant or slowly varying gas temperatures this effect may be important where :

$$\frac{T_m - T_{operation}}{T_m - T_\infty} < \frac{1}{4}$$

Another factor that contributes to the inaccuracy of predicted results, using Equation 37 under low gas temperatures and gas velocities, is that no means is included to account for loss of heat from the sensing element, either link or glass bulb, to the sprinkler frame and piping by conduction. Heskestad and Bill[31] and Ingason[32] have studied means to account for the effects of conduction loss, and measured values for it for sprinkler hardware. Following Heskestad and Bill,[31] Equation 37 can be modified to account for the conduction losses to the frame and piping which are assumed to be at constant temperature equal to the orginal ambient temperature (T_∞) as:

$$t_{\text{operation}} = \frac{\text{RTI}}{\sqrt{U_m}\left(1+\frac{C}{\sqrt{U_m}}\right)}\log_e\left[\frac{T_m - T_\infty}{T_m - T_{\text{operation}} - \frac{C}{\sqrt{U_m}}\left(T_{\text{operation}} - T_\infty\right)}\right] \tag{38}$$

Where C is a conduction loss parameter with units $(m/s)^{1/2}$ obtained by measurement.[31,32] Measured values[31] for the conduction loss parameter range from 0.5 to 1.6 $(m/s)^{1/2}$. Using a value of $C = 0.5(m/s)^{1/2}$ the expected operation time for the sprinkler in the example would increase to 190 sec as predicted using Equation 38, which is over 1 min slower than the prediction of 128 sec (using Equation 37 or Equation 38 with $C = 0$). At a value of $C > 0.74$ $(m/s)^{1/2}$, the heat loss would be great enough to prevent sprinkler operation even though the gas temperature was sustained above the indicated operating temperature of the sprinkler.

All calculations in use today for determining times to operation only consider the convective heating of sensing elements by the hot fire gases. They do not account explicity for any direct heating by radiation from the flames. Research is continuing to evaluate and improve calculations of operation for heat-activated devices, such as sprinklers.

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